International Journal of Mechanical and Production Engineering Research and Development (IJMPERD)
ISSN(P): 2249-6890; ISSN(E): 2249-8001
Vol. 5, Issue 2, Apr 2015, 21-32

CFD ANALYSIS FOR ADAPTABILITY OF PRODUCER GAS FOR POWER GENERATION IN GAS TURBINES

GANGARAJU SRINIVASA SHARMA[1], M. V. S. MURALI KRISHNA[2] & D. N. REDDY[3]

[1]Associate Professor, Department of ME, MVSREC, Hyderabad, Andhra Pradesh, India

[2]Professor, Department of ME, CBIT, Hyderabad, Andhra Pradesh, India

[3]UGC Member, India

ABSTRACT

Gas turbines are low specific fuel consumption prime movers for power generation. The situation of increasing demand for power generation with "Carbon Neutral" technologies is the need of hour. Earlier research studies indicated the feasibility of utilization of low calorific value gas in Gas turbines, but reported a large quantity of Exergy destruction in combustion chamber. The present paper addresses the adaptability of producer gas in a combustion chamber, with a non-premixed model using Ansys-Fluent. A 3D test combustion chamber of 15 cm diameter and 30 cm long with circular air inlets and axially located fuel inlet are modeled and simulations are performed using commercial CFD software. A 25 KW gas turbine combustion chamber is considered with operating pressure of 4 bar absolute working on Producer gas with the fuel composition of 19% H_2, 20 % CO, 2 % CH_4, 10% CO_2 and remaining N_2. Combustion chamber operating with the producer gas is presented. The obtained results are compared with methane and comparative analysis is presented.

KEYWORDS: Flame Stability, Chemical Reaction

INTRODUCTION

Gas turbine combustor design represents an complex task in theoretical, numerical and experimental analysis. Currently, preliminary combustor flow and heat transfer design procedures, which by necessity involve semi-empirical models, are often restricted in their range of application. It was necessary to evaluate the gas turbine combustor for the producer gas, a different fuel from that of the specified liquid fuel. The critical requirements are related to the air-to-fuel ratio, ignition under these conditions, flame stability over the range of operating conditions and establish the inlet operating conditions for the turbine. The simultaneous involvement of evaporation, turbulent mixing, ignition, and chemical reaction in gas turbine combustion is too complex for complete theoretical treatment. Hence large engine manufacturers undertake expensive engine development programs to modify previously established designs through trial-and-error.

Combustor designers without access to proprietary design procedures must derive their own methods from the literature or from experimentation. Numerous published empirical, semi-empirical, and analytical tools have been developed to reduce the need for costly experiments. The two extreme cases, empirical and analytical, differ by the method of derivation. Empirical design tools are correlations derived from experimental datasets whereas analytical ones are discretized versions of the governing equations. Simple empirical correlations provide accurate results quickly and are easily implemented into design codes, yet they are only applicable to cases for which the measured data was based on. Analytical methods, less accurate in comparison to empirical methods, are much more flexible as they are only restricted

by the simplifying assumptions necessary to reduce their complexity and computation time. Hybrid semi-empirical tools combine both empirical and analytical methods to provide a reasonable balance between accuracy and computation time.

LITERATURE REVIEW

As per the earlier research studies made by **Francesco Fantozzi et al** biomass to energy conversion is particularly attractive on the micro scale where internal combustion engines such as micro turbines may be utilized coupled to an indirect gasification system. A RANS analysis has been performed in order to simulate both natural gas and syngas combustion.

Paolo et al have conducted CFD Analysis to study the performance of a modified combustion chamber of a micro gas turbine with the objective to change its fuelling from natural gas to biomass pyrolysis gas. Turbulence in gas turbine combustion chamber is considered important because of its large influence on the combustion characteristics. In order to find optimum conditions in a combustion system, it is essential to get a good approximation of the turbulence. From this viewpoint, multi-dimensional modeling using CFD is pursued and involves simulation with and without combustion. One of the major challenges of CFD in the recent times is that several models need to be combined in order to simulate a complete engine cycle. The k −ε model is the most commonly used turbulence model in CFD, even though its deficiencies are known as indicated by **Versteeg** et al, indeed k −ε model is still considered to the best compromise between computational time and precision.

Kanitkar et al have determined the laminar flame speed (SL) at ambient conditions (0.96 bar, 300 K) by conducting experiments using standard flame tube apparatus for producer gas-air mixture. The gas consisted of 18-23% H2, 17-20% CO, 3-4% CH4, 13-14% CO2 and rest N2. A wide range of mixture ratios were considered within the flammability limits of rich and lean mixtures, namely equivalence ratio _ = 0.47 (26% fuel on volume basis) and 1.65 (56% fuel) for lean and rich limits, respectively. Experimental work at high pressures and temperatures of laminar burning velocity (SL) of producer gas was reported by Keshavamurthy et al. The experiments were conducted in a spherical combustion vessel. Synthetic mixtures of producer gas with a composition of 22% H2, 22% CO, 4% CH4, 10% CO2 and 42% N2 were used to determine SL at initial pressures of 0.5 to 5 bar and ambient temperature. With these initial pressures and temperature a peak pressure of 30 bar during combustion and a maximum unburnt gas temperature of 450 K were obtained. The unburnt gas temperatures were obtained by assuming isentropic compression of unburnt gases as the combustion progresses.

K-ε MODEL

The two dimensional, steady, turbulent, compressible conservation equations were solved using ideal gas approximation of the fluid domain. For steady flow without internal heat sources, the governing equations are continuity, momentum and energy equation.

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0$$

Where ρ is the density, u, v, w are the velocity vectors.

Momentum Equations in Each Direction

Applying the Newton's second law $(force = mass \times acceleration)$ the conservation of momentum equations are given by

X-Momentum Equation:

$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}+\frac{\partial(\rho wu)}{\partial z}=\frac{\partial\sigma_{xx}}{\partial x}+\frac{\partial\tau_{yx}}{\partial y}+\frac{\partial\tau_{zx}}{\partial z}$$

Y-Momentum Equation

$$\frac{\partial(\rho v)}{\partial t}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho wv)}{\partial z}=\frac{\partial\tau_{xy}}{\partial x}+\frac{\partial\sigma_{yy}}{\partial y}+\frac{\partial\tau_{zy}}{\partial z}$$

Z-Momentum Equation:

$$\frac{\partial(\rho w)}{\partial t}+\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}=\frac{\partial\tau_{xz}}{\partial x}+\frac{\partial\tau_{yz}}{\partial y}+\frac{\partial\sigma_{zz}}{\partial z}$$

The stress tensor $\bar{\bar{\tau}}$ is given by

$$\bar{\bar{\tau}}=\mu\left[\left(\nabla.\vec{V}+\nabla.\vec{V}^{T}\right)-\frac{2}{3}\nabla.\vec{V}I\right]$$

Where μ is the molecular viscosity, I is the unit tensor, and the second term on the right hand side is the effect of volume dilation.

Energy Equation

Energy is neither created nor destroyed. It is always conserved.

$$\frac{\partial(\rho E)}{\partial t}+\frac{\partial(\rho uE)}{\partial x}+\frac{\partial(\rho vE)}{\partial y}+\frac{\partial(\rho wE)}{\partial z}$$
$$=\frac{\partial\left(u\sigma_{xx}+v\tau_{xy}+w\tau_{xz}\right)}{\partial x}+\frac{\partial\left(u\tau_{yx}+v\sigma_{xy}+w\tau_{yz}\right)}{\partial y}$$
$$+\frac{\partial\left(u\tau_{zx}+v\tau_{zy}+w\sigma_{zz}\right)}{\partial z}+\frac{\partial\left(k\frac{\partial T}{\partial x}\right)}{\partial x}+\frac{\partial\left(k\frac{\partial T}{\partial y}\right)}{\partial y}+\frac{\partial\left(k\frac{\partial T}{\partial z}\right)}{\partial z}$$

In the above equation, $E=h-\frac{p}{\rho}+\frac{v^2}{2}$, Where sensible enthalpy h is defined for ideal gases as $h=\sum_j Y_j h_j$ and for incompressible flows as $h=\sum_j Y_j h_j+\frac{p}{\rho}$, Y_j in the above expression is the mass fraction of species j and $h_j=\int_{T_{ref}}^{T} c_{p,j}dT$, where T_{ref} is the reference temperature and its value is 298.15 K.

RANS Equation

$$\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\,\delta_{ij}\,\frac{\partial u_i}{\partial x_i}\right)\right]+\frac{\partial}{\partial x_j}\left(-\rho\overline{u'_i u'_j}\right)$$

The left hand side of this equation represents the change in mean momentum of fluid element owing to the unsteadiness in the mean flow and the convection by the mean flow. This change is balanced by the mean body force, the isotropic stress owing to the mean pressure field, the viscous stresses, and apparent stress $\left(-\rho\overline{u'_i u'_j}\right)$ owing to the fluctuating velocity field, generally referred to as the Reynolds stress. These equations form a set of coupled, nonlinear partial differential equations. It is not possible to solve these equations analytically for most engineering problems. However, it is possible to obtain approximate computer-based solutions to the governing equations. The standard $k-\epsilon$ model is a semi-empirical model based on model transport equations for the turbulence kinetic energy (*k*) and its dissipation rate (ε). The model transport equation for *k* is derived from the exact equation, while the model transport equation for ε was obtained using physical reasoning and bears little resemblance to its mathematically exact counterpart.

Computational model

A 3-d combustion chamber was designed using Ansys Workbench in design mode and the dimensions of the chamber are represented in the figure 1. Grid generation represents a tedious task for CFD analysis. It is a time-consuming task and, in spite of steady advances in automatic mesh generation, it still requires the skill of a CFD practitioner to yield a suitable mesh. The choice of the type of grid depends on geometrical complexity and on physics. For gas simulation a producer gas-air mixture was used with chemical composition of 20% CO, 19% H_2 ,2% CH_4, 10% CO_2 and remaining N_2.

Calorific value of Producer gas is estimated by earlier authors and it is indicated as about 3000 Kcal/kg[]. Heat transfer coefficient of 20 w/m^2K, ambient Temperature of 298 K, P_{ata} =1.01325 bar and density of air is 1.125 kg/m^3. The thermal properties of the producer gas and species are function of temperature.

Table 1

Gas Turbine Standard Operating Parameters	Conditions
Ambient Pressure(bar)	1.0325
Ambient Temperature(k)	298
Delivery Pressure of compressor (bar)	4
Isentropic compressor Efficiency	90%
Isentropic turbine Efficiency	90%
LCV of Producer gas (KJ/Kg)	6000
Compressor Exit temperature(K)	456
Turbine Inlet temperature(K)	925
Turbine outlet temperature(K)	656
Thermal Efficiency	23.60%
Mass flow rate of fuel(kg/sec)	1.39e-02
Mass flow rate of primary air (kg/sec)	1.53e-02

Non premixed combustion model with turbulence (K-ε model with standard wall functions) is considered with the composition of gas having 19% H_2 , 20 % CO, 2 % CH_4 , 10% CO_2 and remaining N_2 is considered. The combustion chamber is of 30 cm length and 15 cm diameter with fuel inlet at a diameter of 5 cm and 12 circular air inlets of diameter 1.25 cm. Air of mass flow rate equal to1.53e-02 kg/sec is admitted and 1.39e-02 kg/sec of fuel is admitted through fuel inlet system. Pressure in the combustion chamber is maintained at 4 bar absolute. It is observed that there is about 0.08 bar pressure drop in the combustion chamber. The outlet is at about 3.92 bar with no back pressure. The geometry of combustion chamber test specimen is indicated in the figure 1. The figure 2 indicates the meshed geometry and figure 4 indicates the plots of residuals after convergence.

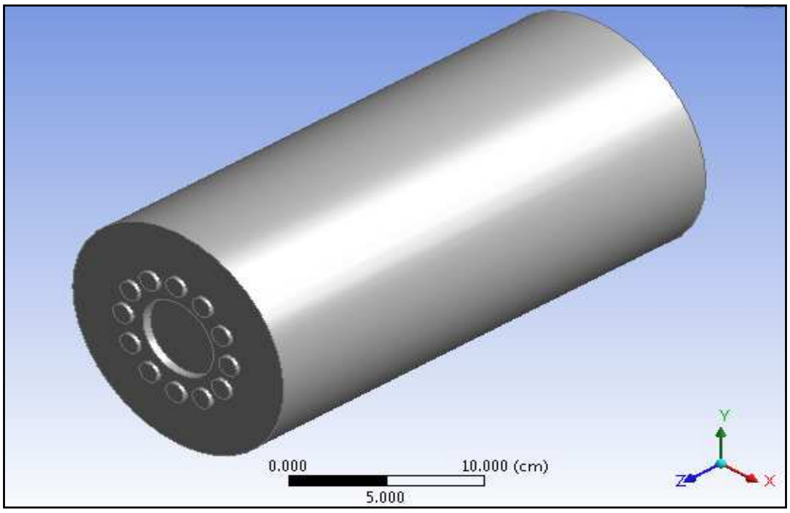

Figure 1: Combustion Chamber

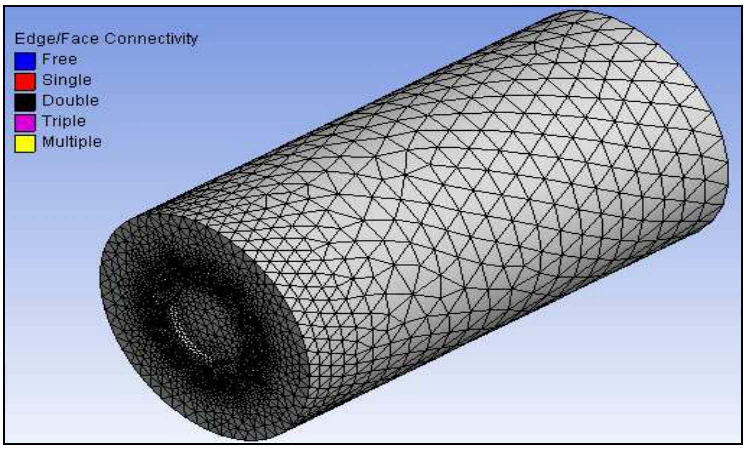

Figure 2: Meshed Geometry

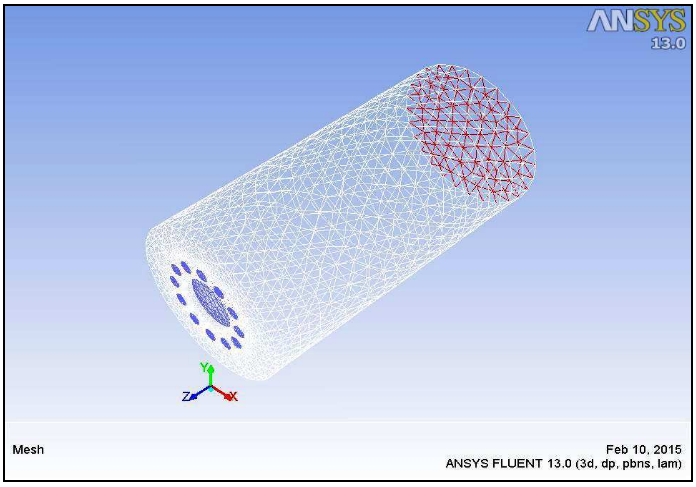

Figure 3: Meshed Geometry with Boundary Conditions

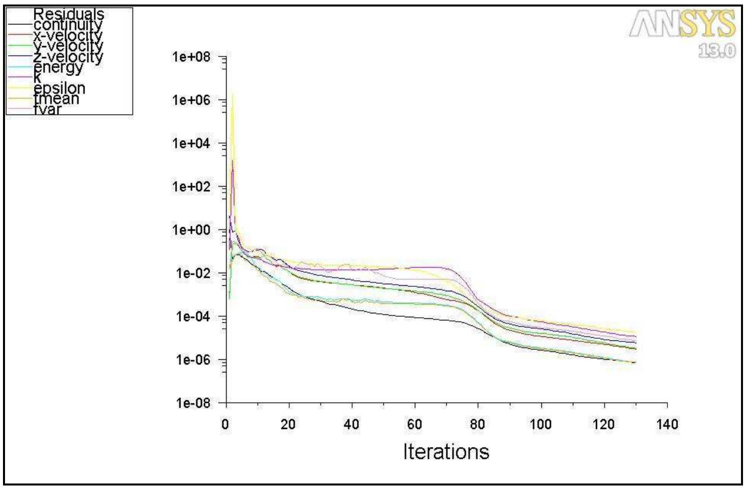

Figure 4: Residuals Plot

RESULTS AND DISCUSSIONS

With air fuel ratio of 1.1, the temperature distribution and velocity distributions at various planes of 5cm,10 cm and 20 cm from the fuel inlet is indicated in the figure 5 and 6. It is observed that temperature is uniformly distributed with

average temperature of 1025 K and average velocity of 1.93 m/sec. The maximum temperature of about 1500 K is attained at a distance of 10cm from the fuel inlet where outer surface is heated, whereas the inner core is about 900 K. The combustion chamber is capable of generating 25 KW of electric power. The air fuel ratio is varied from 0.8 to 1.2 and temperature, velocity and pressure plots are obtained. The pressure distribution along the axis of the combustion chamber is indicated in the figure 9 and drop of 0.08 bar of total pressure is observed. The velocity of producer gas mixture leaving the combustion chamber is observed to be about 3.46m/sec whereas the methane mixture is leaving with a velocity of about 1.73 m/sec. Hence there is a chance of flame entering the post combustion region which is not recommended for gas turbine operation. Hence secondary air has to be mixed with the flame at the locations of 5cm, 10 cm, 15 cm and 20 cm from the fuel inlet and flame has to be quenched.

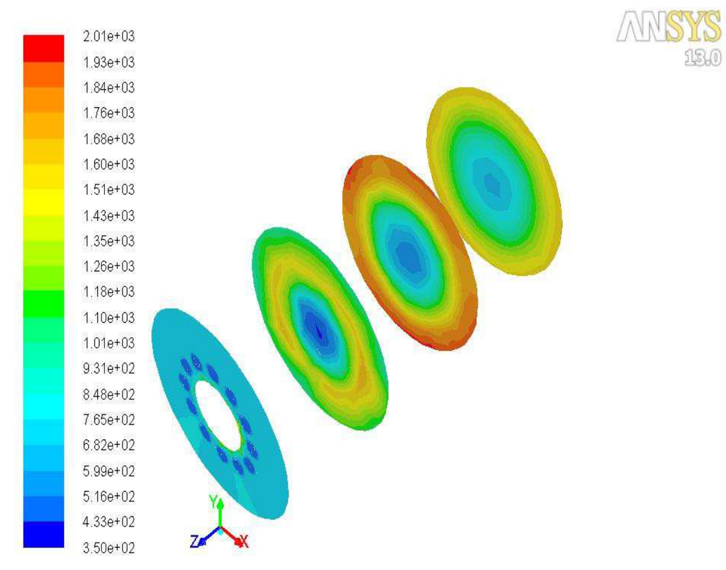

Figure 5: Temperature distribution Plots at 0, 5cm, 10 cm and 20 cm from the Fuel Inlet

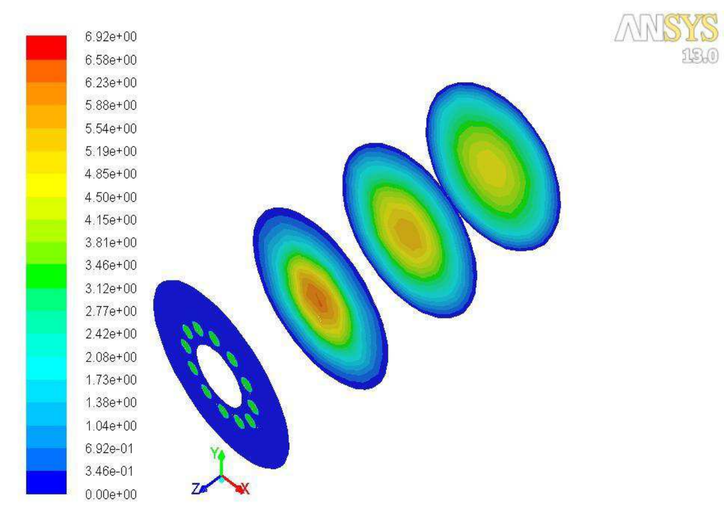

Figure 6: Velocity Distribution Plots at 0, 5cm, 10 cm and 20 cm from the Fuel Inlet

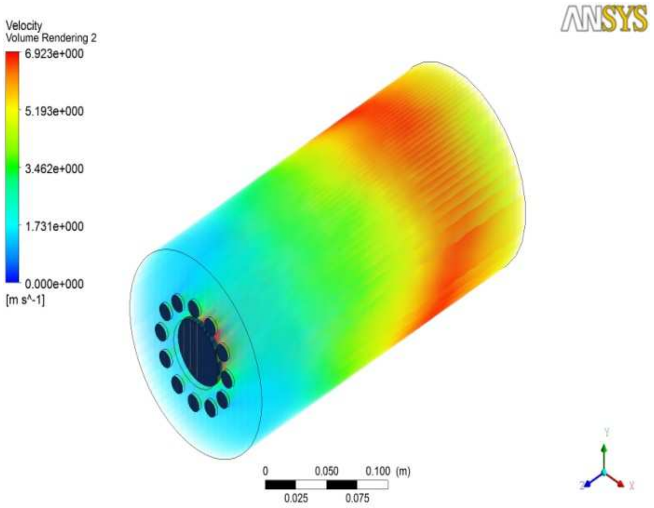

Figure 7: Velocity Contour in Combustion Chamber

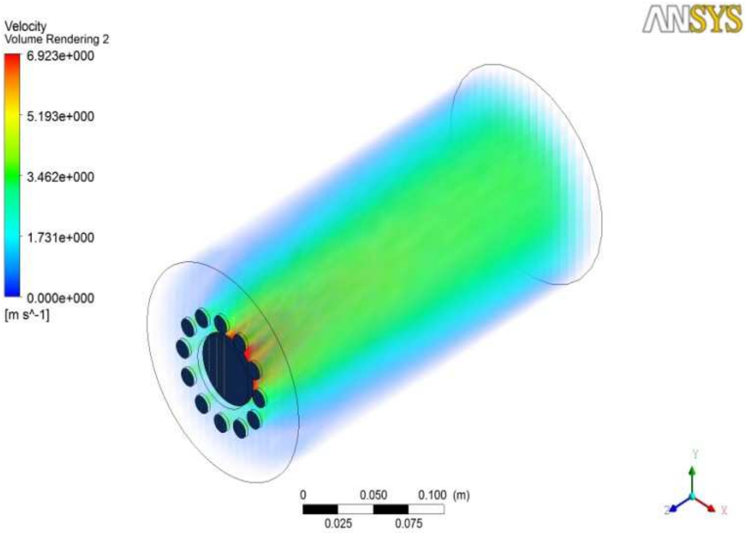

Figure 8: Temperature Contour in Combustion Chamber

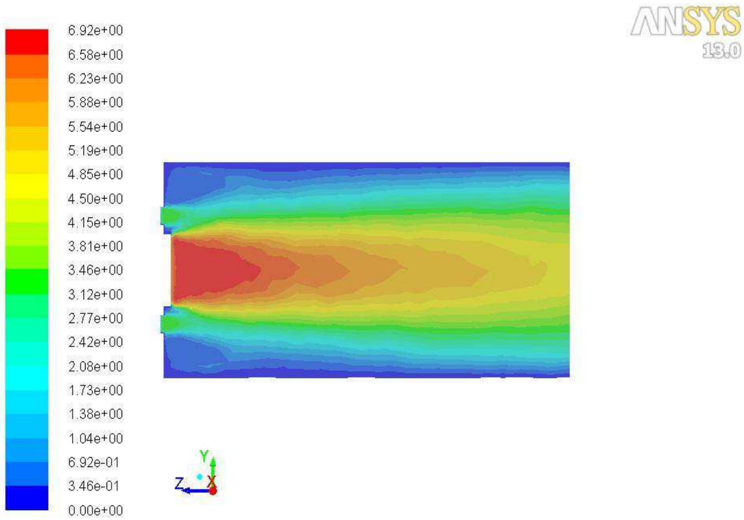

Figure 9: Velocity along Axial Direction

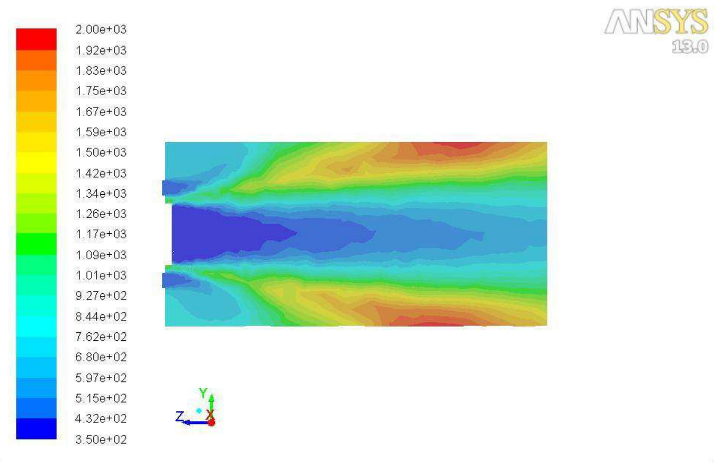

Figure 10: Temperature along Axial Direction

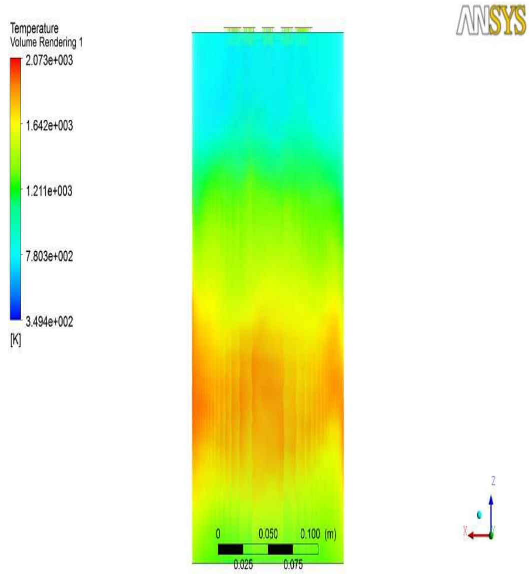

Figure 11 : Temperature Distribution with Methane as Fuel

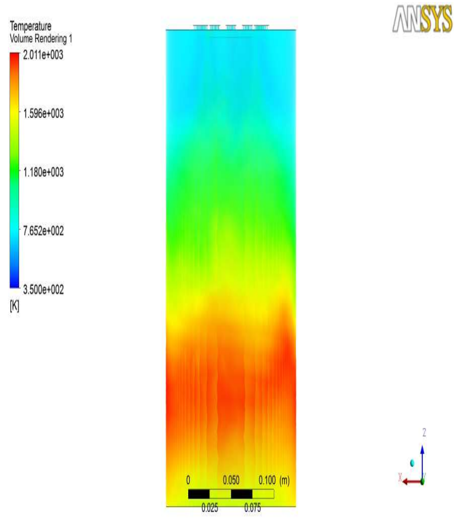

Figure 12: Temperature Distribution with Producer Gas as Fuel

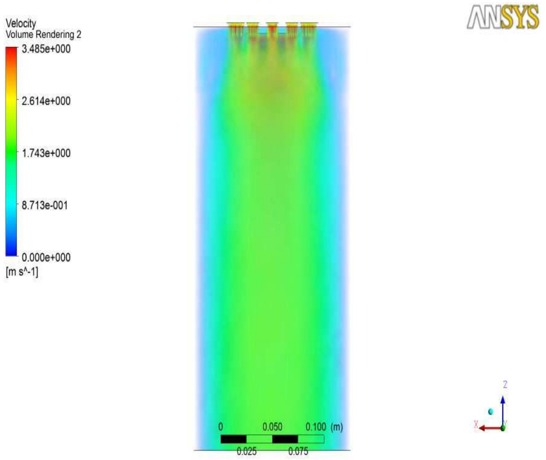

Figure 13: Velocity Distribution Using Methane as Fuel

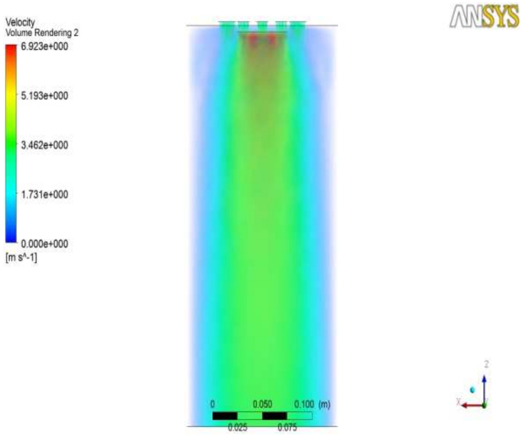

Figure 14: Velocity Distribution Using Producer Gas

CONCLUSIONS

The gas turbine working on methane is working with high calorific value fuel approximately about 22500 KJ/Kg, with an A/F ratio of 12:1. Whereas gas turbine working with producer gas is working on LCV gas having calorific value of 6 KJ/Kg with an A/F ratio of 1.2. Though the calorific value is less the energy density for same power generation is more in case of producer gas than methane. The Turbine inlet temperatures at gas turbine conditions are within the specified and recommended range of OEM's of gas turbines, which prevents the flame burnout inside the expander. The flame velocities are higher in case of producer gas than in methane along axial direction and it is observed that the exit velocity is of the order of 3.46 m/sec. The temperature attained at the distance of 15 cm to 20 cm from the fuel injection plane is more in case of producer gas. Hence dilution of lame in this region is more necessary and secondary air of about 55% higher than primary has to be provided to reduce the velocity and obtain equal temperature distribution. The fluid flow analysis coupled with chemical kinetics has indicated that producer gas can be used as the alternate fuel for power generation in Biomass integrated gasification/Gas turbine Technology for power generation, with average Lower calorific value of product gas about 6MJ/kg.

ACKNOWLEDGEMENTS

We would like to sincerely acknowledge the support and motivation provided by director, Center for Energy Technology, Osmania University for providing facilities for conducting research on biomass gasification and testing the gas.

REFERENCES

1. P. J. Paul and H. S. (Editors) Mukunda. Recent advances in biomass gasification and combustion: Proc. of 4th meet on Biomass Gasi_cation and combustion at Mysore,6-8th Jan. 1993. Interline publishing, Bangalore, India, 1993.
2. G. Sridhar, P. J. Paul, and H. S. Mukunda. Biomass derived producer gas as a reciprocating engine fuel-an experimental analysis. Biomass and Bioenergy, 21:61{72,2001.
3. G. Sridhar. Experiments and modelling studies of producer gas based spark-ignited reciprocating engines. PhD Thesis, CGPL, IISc, 2003.
4. K. N. Lakshmisha. Computational studies on the flamability limits of premixed gases. PhD Thesis, IISc, 1991.
5. C. J. Rallis and A. M. Garforth. The determination of laminar burning velocity. Progress
6. in Energy and Combustion Science, 6:303{329, 1980.
7. D.V. Sridharan. Gasifiers once solved energy problems - they might, yet again, available at. http://www.goodnewsindia.com/Pages/content/discovery/cgpl.html, December 2001.
8. Shashikantha, P. K. Banerjee, G. S. Khairnar, P. P. Kamat, and P. P. Parikh. Development and performance analysis of a 15 kwe producer gas operated si engine. Proc. Of 4th meet on Biomass Gasification and combustion at Mysore,6-8th Jan. 1993 Interline publishing, Bangalore, India, 4:219{231, 1993.
9. A.V. Bridgewater, Renewable fuels and chemicals by thermal processing of biomass, Chemical Engineering Journal, 91 (2003) 87-102.